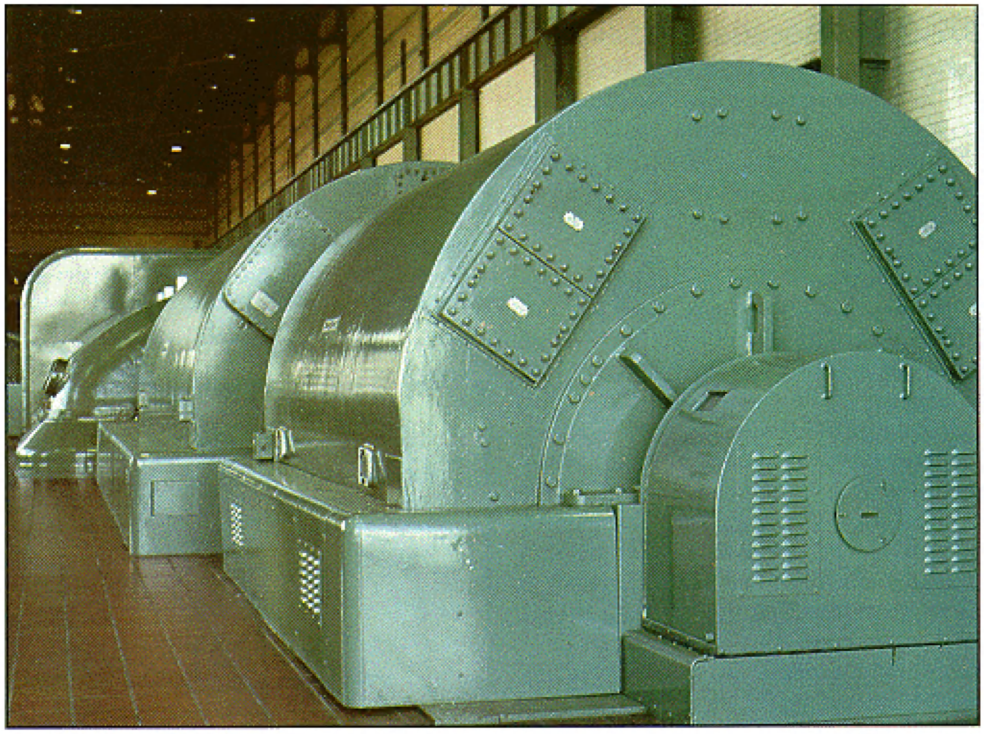
Anatomy of a shaft crack



The analytical methods used by PSEG enabled their engineers to properly diagnose the shaft crack on the 30-year-old steam turbine generator.

When Hugh Stott, Steam Turbine Engineer for Public Service Electric and Gas (PSEG) received the report of high vibration on Unit No. 7 steam turbine generator at the Burlington, New Jersey power station, he knew the problem was serious.

Lou Corbett, Burlington Chief Engineer, had called that morning to tell of observations of higher than normal critical speed and on-line vibration. Station operations personnel had observed a trend of increasing vibration amplitude during startups since the intermediate pressure (IP) and double flow low pressure (LPDF) rotors were balanced on the 30-year-old, 185-megawatt (MW) unit two months before.

Now, Corbett reported that the 1X vibration amplitude on Bearing No. 6 had increased to 6.7 mils peak-to-peak after the startup.

Stott immediately telephoned Michael Wallo, Principal Staff Engineer of PSEG's Research and Testing Laboratory and requested him to monitor the vibration amplitude and phase angle levels on Unit No. 7. The vibration information was needed to determine the cause of the problem.

As it turned out, this vibration data and the historical vibration information compiled by the Research and Testing Laboratory over the past six months was crucial in diagnosing that the problem was a shaft crack.

Initially, however, Stott did not suspect a shaft crack. Imbalance, rubbing, or other influences were the other primary suspects.

The analytical methods employed by PSEG enabled Stott and PSEG's engineering staff to sort out the symptoms and properly diagnose the problem. The analytical methods were based on past experience, interviews with station operations personnel, and recent training on shaft crack detection using vibration analysis.

Twenty-three years as a turbine engineer, 18 with General Electric and five with PSEG, had taught Stott that diagnosing machine problems was much like conducting a police investigation.

"You need as much information as possible to determine what's wrong when you have a running problem with a machine," Stott explains. "And in evaluating the information, you have to separate the facts from opinion. You must stick to the facts."

Stott and PSEG's Research and Testing Laboratory staff began gathering the facts. After requesting additional vibration data from the laboratory, Stott interviewed station operations personnel to learn what they had observed during the Unit No. 7 startup.

Meanwhile, the Research and Testing Laboratory had brought data recording equipment to the Burlington station. They recorded Keyphasor, proximity, and seismic transducer signals through the monitor outputs. The data was later reduced in the mobile unit using a Bently Nevada ADRE® (Automated Diagnostics for Rotating Equipment) with a Bently Nevada Multichannel Vector Filter (VF-M) and spectrum analyzer.

The vibration data indicated significant 1X and 2X vibration amplitude levels on Bearing Nos. 5 and 6 at a load of 72 MW and a main generator field of 1,400 amperes. On Bearing No. 5, the 1X vibration amplitude was 6.0 mils and the 2X vibration amplitude was 1.0 mil. On Bearing No. 6, the 1X vibration amplitude was 6.7 mils and 2X vibration amplitude was 1.5 mils.

The 1X vibration amplitude levels were compared with the historical data. Based on the comparison, the machine was run up again to acquire 2X vibration amplitude data for further comparison. The turbine generator reached 3,048 rpm before danger alarms were triggered for excessive vibration levels at the IP, LPDF, and generator rotors.

Runup data was recorded on a 16-channel tape recorder. The Research and Testing Laboratory's mobile unit was being used at another station 60 miles away. The recorded data was driven to the station and reduced.

Robert Konnyu of the Research and Testing Laboratory called Stott with the

results just as Unit No. 7 was being run up again. The results—high 1X and 2X vibration levels—confirmed that the machine's stability had decreased to the point where it could be hazardous to continue operation.

The machine was run down, and vibration data was recorded and reduced by the Research and Testing Laboratory. High ½X, 1X, and 2X vibration amplitude levels were noted on Bearing Nos. 3, 4, 5, and 6:

Bearing No.	Peak-to-Peak ½X	Vibration 1X	Amplitude 2X
3	9.0 mils	7.4 mils	——
4	5.5 mils	4.6 mils	1.5 mils
5	3.5 mils	2.8 mils	6.0 mils
6	1.0 mil	6.3 mils	8.0 mils

The high 1X and 2X vibration amplitude levels caused Stott to suspect a shaft crack. Stott and Konnyu had recently attended a Bently Nevada seminar where results of a laboratory study on shaft cracks were presented.

The study results indicated that the modification of the shaft stiffness by the crack causes changes in the dynamic response to major excitation forces: inertial force due to unbalance and the gravity force. The largest differences in the shaft vibration response amplitudes were observed in the 1X and 2X regions of frequency.[1]

To determine whether the problem was a shaft crack or another malfunction, Stott compared the 1X vibration amplitude data from the previous day's rundown with the historical vibration data archived by the Research and Testing Laboratory from a rundown four months before (see Table 1). Comparison showed that the 1X vibration levels had increased significantly on Bearing Nos. 1, 3, 4, 5, and 6.

The machine was not put back in service, but opened and inspected. Inspection revealed a crack 400 degrees around the circumference with a 270-degree penetration to the bore in the double flow low pressure rotor at the generator end in the wheel-to-shaft radius.

The Research and Testing Laboratory's evaluation following the inspection concurred with the prediscovery of the shaft crack, which affected the vibration levels. The laboratory compared the machine's synchronous amplification factor, damping ratio, and spring index from the rundown six months before to the rundown on the first day of this case.

The synchronous amplification factors had increased substantially on Bearing Nos. 5 and 6:

Bearing No.	Synchronous Amplification Aug.	Factor Dec.
5	8.42	16.94
6	4.75	8.09

The synchronous amplification factor measures the susceptibility of a rotor to vibration when rotational speed is equal to the rotor natural frequency. Generally, a synchronous amplification factor above 5 or 6 indicates low damping and less machine stability.

Comparison of the damping ratio and spring index also indicated less machine stability. The generator end of the LPDF rotor had very little damping compared to the turbine end of the same rotor. Bearing No. 5 had a damping ratio of 2.95 x 10^{-2}. Bearing No. 4 had a damping ratio of 8.06 x 10^{-2}.

The damping ratio shows the effect of damping to the response of the rotor at its natural frequency. The major effect of damping is on the machine's synchronous amplification and phase response at resonance.

The spring index on Bearing No. 5 of the LPDF rotor had decreased from 4.4 x 10^6 lb./in. to 3.63 x 10^6 lb./in. The spring index, or stiffness, is the amount of deflection of the rotor and is measured in lb./in.

Conclusion

The historical vibration data, plus ability of PSEG's Research and Testing Laboratory to acquire and reduce vibration data in a short period of time, was crucial in providing Stott and PSEG's engineering staff with the information needed to correctly diagnose the problem on Burlington's Unit No. 7. Training on data interpretation methods also played a major role in sorting out the symptoms and diagnosing the shaft crack.

This disciplined approach aided Stott in making a sound decision to shut down the machine prior to catastrophic damage being incurred. After major repairs to the LPDF rotor, Burlington's Unit No. 7 was returned to service in June 1985. The rotor was repaired by cutting off the rotor between the generator end L-0 and L-1 stages, welding on a new forging, machining it to original dimensions, and reassembling the L-0 blades. The rotor is running well with low levels of vibration.

[1] *Muszynska, Agnes, "Shaft Crack Detection," Bently Rotor Dynamics Research Corporation, 1982.*

TABLE NO. 1
Transient Data — Absolute Values[1]

Brg. No.	Rundown Data 24 Aug. 84 Speed (rpm)	1/2X (mils)	1X (mils)	2X (mils)	Rundown Data 3 Dec. 84 Speed (rpm)	1/2X (mils)	1X (mils)	2X (mils)	Runup Data 4 Dec. 84 Speed (rpm)	1/2X (mils)	1X (mils)	2X (mils)
1	3100	—	0.1	—	3200	0.5	2.2	—	3036	—	2.8	0.4
									3048	—	3.3	0.7
3	3100	—	2.8	—	3200	9.0	4.6[2]	—	3036	—	8.6	1.0
									3048	—	10.6	1.8
4	3100	—	1.5	—	3050	5.5	4.6	1.5	3023	—	6.3	1.5
									3048	—	8.4	1.1
5	3100	—	3.4	—	3050	3.5	2.8	6.0	3023	—	5.5	4.0
									3048	—	5.7	3.5
6	3100	—	2.2	0.7	2970	1.0	6.3	8.0	3033	—	6.8	6.3
									3048	—	7.1	6.9
7	3100	—	3.0	—	2970	—	1.8	2.5	3033	—	1.4	2.0
									3048	—	1.4	2.3

[1] Uncompensated data taken from cascade plots

[2] At 3231 rpm — 7.4 mils @ 1X (not shown on plot)